AN EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER AND PRESSURE DROP IN PARALLEL ROD ARRAYS

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CERTIFICATE

This is to certify that this work has been carried out under my supervision and has not been submitted elsewhere for a degree.

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TO MY PARENTS

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ABSTRACT

Experimental investigations on axial flow heat transfer, temperature distribution and pressure drop were carried out in closely packed, 19 rod delta arrays at pitch to diameter ratios of 1.1 and 1.2. Air was the cooling medium. The test rods were internally heated by commercial resistance heaters. Considerable variation was observed in the surface temperatures and heat transfer rates from rod to rod. The circumferential variation of rod surface temperature was clearly discernible, though not of a high order. The effect of the finiteness of the array was clearly evident in the form of the distortion of the temperature field near the periphery. This distortion effect was found to penetrate into the second ring of rods also. A good degree of interchannel mixing was revealed by a study of temperature plots. It was also found that both heat transfer co-efficients and friction factors lie below those for circular tubes for the array with pitch to diameter ratio of 1.1 and above those for circular tubes for the array with pitch to diameter ratio of 1.2. Experiments could be conducted upto a Reynolds number of about 11,300.

LIST OF ILLUSTRATIONS

LIST OF ABBREVIATIONS AND SYMBOLS

c_p	-	Specific heat of air, at constant pressure.
d	-	Diameter of test rod.
d_e	-	Hydraulic equivalent diameter of channel surrounding an individual rod.
$\bar{h}$	-	Average film heat transfer coefficient
h_w	-	Pressure difference across orificemeter, inches of water
k	-	Thermal conductivity of air.
m	-	Mass flow rate of air in an individual channel.
p	-	Pitch of test rods
$\bar{q}$	-	Average heat transfer rate
$t_{m,in}$	-	Air inlet, Mean mixed temperature.
$t_{s,x}$	-	Rod surface temperature at distance x from the beginning of the heated portion.
μ	-	Absolute Viscosity of air in centipoise.
A	-	Total heat transfer area of test rods.
D_2	-	Orificemeter diameter.
E	-	Voltage drop across rod heater.
K	-	Orifice discharge constant.
$\hat{M}$	-	Mass flow rate in the bundle as a whole.
P	-	Pressure
Q	-	Volume rate of flow of air

R_{res} - Rod heater resistance.

R - Universal Gas constant.

V - Fluid velocity.

W - Total air mass flow rate.

X - Total heated length of test rod.

Nu - Nusselt number.

Pr - Prandtl number.

Re - Reynolds number.

β - Diameter ratio.

γ - Specific heat ratio.

ρ - Mass density of air.

CHAPTER I

INTRODUCTION

Arrays of parallel tubes have long been used extensively in shell and tube cross flow heat exchangers. In the development of gas cooled reactors capable of yielding high gas temperatures, this geometry, with parallel flow, is receiving increased interest. This geometry has been suggested with a view to providing increased heat transfer surface in order to compensate for reduced heat transfer co-efficients associated with gas cooling. This type of fuel configuration also leads to high conversion ratios in thermal reactors.[17] In this case the fuel element configuration is arrived at by considerations of obtaining a large fast effect in U^{238} without drastically reducing the heat transfer rate. These considerations lead to closely packed rod cluster.

The thermal characteristics (Heat transfer rate, temperature distribution) of this configuration are complicated and have not yet been fully understood. The complications arise from the multiplicity of boundary conditions which may be imposed. Also the shear stress distribution in the flow is quite different from that found in ordinary ducts like parallel plates and circular tubes and may give rise to heat transfer rates within the flow

passage which cannot be easily predicted. The varying shape of the flow channel surrounding the individual rods causes peripheral variation in heat transfer rates and temperature distribution.[2,4,7,12,14] Especially for gaseous coolants heat transfer rates will be one to two orders of magnitude less than that for water or liquid metals. This causes increased circumferential variation in temperature distribution and hence problems of rod bowing will be magnified.

So far most of the analyses, both theoretical and experimental, have been done for the case of infinite or simulated infinite arrays. In practice clusters of 7, 19, 21, ... rods are used. The finiteness of the array is another element of complexity which has not been studied. Most of the analytical models assume the absence of interchannel mixing. But for finite arrays this may not be valid.

In the present investigation, an attempt has been made to study, experimentally, the following problems.

(i) Effect of finiteness of the array on the temperature distribution.

(ii) The extent to which boundary effects penetrate into the array.

(iii) To verify, qualitatively, the assumption regarding the absence of interchannel mixing.

(iv) The effect of change in spacing of rods on heat transfer rates and how these rates compare with that given by Colburn's relation for circular tubes.

(v) Determination of friction factors for different spacings of the rods and their agreement with the values for turbulent flow in smooth pipes.

Answers to above problems has been sought to be obtained by conducting experiments in a 19 rod, triangular arrays with pitch to diameter ratios of 1.1 and 1.2. The temperature fields of different rods in the array in experiments proposed to be conducted by heating different assortment of rods, are expected to throw light on finiteness of the array, the extent to which the boundary effects penetrate into the array and the assumption regarding interchannel mixing. It will also be possible to find the effect of varying pitch to diameter ratio on heat transfer coefficients and friction factors.

CHAPTER II

A REVIEW OF PREVIOUS RELATED WORK

A. Analytical Approach:

The techniques for analytical treatment of turbulent flow heat transfer have been developed to the extent where fairly accurate predictions can be made in simple geometries. Although there are uncertainties about the basic mechanisms of turbulent flow, a fairly accurate analysis is possible from the macroscopic point of view by using the concept of turbulent eddy diffusivity. Sufficient experimental data is available on velocity distribution in turbulent boundary layers and in circular ducts to permit calculation of the values of eddy diffusivity for momentum and eddy diffusivity for heat. These data have been successfully applied to the solution of the energy equation for simple flow geometries like parallel plates and circular tubes, but are not so easily applied for the case of more complex flow geometries.

Although the geometry of circular annulus is axisymmetric the simple extension of the analysis for annulus to this geometry cannot be easily accomplished. In contrast to the circular tube case, the shear stress distribution in the annulus is not linear with distance from the wall, and the location of zero shear plane depends on the ratio of the inner to outer radii.[1]

Kays and Leung[8] made the first successful attempt to obtain analytical solution of heat transfer in the circular annulus by using velocity measurements made in annuli of various ratios of inner to outer radii. The energy equation was solved for fundamental solutions by making use of the location of the zero shear plane, the eddy diffusivity for momentum and the velocity profile obtained from the above data.

The annulus problem leads to the conclusion that the universal turbulent velocity profile, or "the law of the wall", is not applicable in all flow geometries. Hence it is necessary to have velocity measurements for each geometry of interest over a range of geometries.

Deissler and Taylor[1] analyzed turbulent flow in a circular annulus by assuming that the universal turbulent velocity profile holds for this geometry in the same manner as it holds for the circular geometry. This assumption is based on their argument that the velocity distribution normal to the wall is independent of the shear stress distributions. They obtained the solution by graphically fitting the velocity profile to the flow geometry. Similarly non-dimensional temperature profile is graphically fitted to find the resulting temperature field for uniform heating. They have extended this kind of analysis to parallel rod bundle.[2] Their analysis shows the existence of considerable peripheral variations of wall temperature as well as reduced heat transfer for closer rod spacings.

An analytical approach has been made by Maresca and Dwyer[10] to the turbulent flow by considering an individual rod in a parallel rod array to be analogous to the centre tube of a concentric circular annulus. As seen in figure No. 1 the zero shear planes define a hexagonal flow passage which may be replaced by a circular tube of equivalent flow area. This flow passage, bounded by the rod and the zero shear plane, is assumed to be equivalent to the region between the centre tube and the zero shear of an annulus. The advantage of this model is that the velocity field in the annulus and the equivalent annulus model are identical. The main limitation is how well the equivalent circular zero shear plane will represent the actual hexagonal shape - an approximation which becomes invalid as the rods are spaced closely. This and similar type of analysis made by Dwyer and Tu[4] make use of experimental velocity profiles in circular tubes for calculating the eddy diffusivities.

Reynolds, Lundberg and McCuen[13] developed an analytical solution to the circular tube annulus by the technique of superposition of fundamental solutions. The energy equation is made linear and homogeneous in temperature by making suitable simplifying assumptions and idealisations. Any linear combination of solutions will also be a solution. This allows the superposition technique to be used. The fundamental solutions are obtained either analytically or experimentally. Kays and Leung[8] made a similar analysis for the case of turbulent flow in a circular

tube annulus. Sutherland and Kays extended the technique to the case of an infinite parallel rod array. They obtained fundamental solutions experimentally. Sutherland and Kays[14] defined their fundamental solution as the expression for the temperature on the rod numbers 1, 2 or 3 (Figure 3) bounding the unit flow passage, resulting from a uniform heat flux on one of these, zero heat flux from the other two and no heat diffusion through the gaps between the rods. The fundamental solutions were obtained experimentally, by measuring the wall temperatures when (say) Rods 1 and 6 were heated uniformly. Under these conditions, all the four unit flow passages A_{123}, A_{134}, A_{126} and A_{236} have symmetrical temperatures distributions and there cannot be any energy diffusion across the boundaries.

In most analytical models it is assumed that there is no mixing across zero shear boundaries. The plane of zero shear implies no momentum exchange, but there is a finite eddy diffusivity which can transfer thermal energy to adjacent channels. In many cases, however, this may be ignored. This is justifiable in an infinite array. But in the case of arrays of 7, 19, 31 rods which are more commonly used energy transfer between adjacent subchannels may be significant.

B. Experiments in Parallel Rod Arrays:

Experimental investigations of heat transfer in parallel rod geometry have been reported over the past 10 years. One of the earliest was an experimental investigation of Miller, Brynes and Benforado[11] who reported data for an infinite (simulated with dummy i.e., unheated rods) array of rods in a delta pattern at P/d = 1.462. These tests, with water as the working fluid, covered a range of Reynolds numbers from 70,000 to 700,000 and Prandtl numbers from 1.10 to 2.75. Water was circulated at a maximum rate of 20 ft/sec. The rods were left unheated and instead the water was heated. Although the test section was 4 ft. long, the heat transfer measurements were made with a 4 inch heated length at the mid section of one rod. Relatively high heat fluxes of 50,000 to 200,000 Btu/sq.ft./hr were obtained. They concluded from these tests that the heat transfer coefficients measured were 40 percent higher than those predicted by circular tube correlation of Colburn.

$$Nu = 0.023 \ Re^{0.8} \ Pr^{1/3}$$

Eckert[5] suggested the hydraulic equivalent diameter (= 4 x flow area/ Wetted parameter) as the linear dimension. They did not find any circumferential variation in rod surface temperature.

Dingle and Chastain[3] tested both delta and square patterns at pitch to diameter ratios (p/d) of 1.12, 1.20 and

1.27 with water in a uniformly heated nine - rod array. The Prandtl number ranged from 1.18 to 1.78. It was concluded from their experiments that the heat transfer coefficient could be correlated adequately by Colburn's relation for circular tubes, although the data for larger spacings tend to lie about 15 percent higher. There was no variation of wall temperature around the periphery of the rod.

In the 1961 International Heat Transfer Conference at Boulder, Colorado lot of work has been reported on parallel rod geometries particularly as applied to nuclear reactor fuel elements. Palmer and Swanson[12] investigated a very close spacing, p/d = 1.015, using a large scale seven - rod array with dummy rods at the wall to simulate an infinite array. Their tests with air covered the Reynold's number range from 10,000 to 60,000. Thick walled aluminium tubes were used as test rods, internally heated by electrical resistance heaters. This was critisised during the discussion, since represented neither a constant temperature nor a constant heat flux boundary condition. Instead of measuring circumferential temperature variation, they measured the duration in heat transfer coefficient directly, by a balancing technique. A strip of temperature controlled surface heater was fixed in the central rod of the array. The local circumferential variation of the heat flux were determined from measurements of the electrical power necessary for

equalising the temperature of the controlled heater surface to that of the adjacent tube wall, while the central rod was rotated about its longitudinal axis. From their experiments it was concluded that heat transfer rates compared favourably with those given by Colburn correlation. The peripheral variation of local heat transfer coefficient was observed to be 3 times (i.e., maximum to minimum ratio). They also found that fully developed flow conditions were established with a length of 36 equivalent diameters at inlet.

Hoffman, Wantland and Stelzman[7] reported data on a 7 rod cluster of P/d = 1.715. The rods were heated by passing an alternating current. Tube surface temperatures were measured by means of movable thermocouple probes, inserted into the tubes. The circumferential temperature variation was significantly less than (about 5 percent, as against 25 percent) that predicted by Deissler and Taylor.[2] The average heat transfer co-efficients exceeded that given by Colburn equation by about 20 percent. Friedland and Bonilla[6] have reported about the experiments conducted with Mercury flowing in parallel rod arrays. The results obtained by them showed that:

1. the circumferential variation in temperature was of a small order.
2. Sufficient bulk temperature variations could exist, resulting in the Nusselt numbers varying by a factor of 4 between different tubes in the array.

3. Entrance effects were negligible after 35 equivalent diameters.

Sutherland and Kays investigated rod arrays for $p/d = 1.0, 1.5, 1.25$, with air as the cooling medium. Measurements were made with one rod heated alone, six rods in a ring heated and all rods in the array equally heated. It was found that the heat transfer co-efficient was substantially less than that predicted for circular tube for $p/d = 1.0$, and somewhat greater than the circular tube for $p/d = 1.15$ and 1.25. The surface temperature around the rod was also found to vary, but the percent variation was not large.

There are some reports available on the pressure drop associated with parallel rod geometries. LeTourneau, Grimble and Zerbe, Subbotin, Ushakov and Gabrianovich have conducted investigation to determine friction factor.

LeTourneau, Grimble and Zerbe[18] determined friction factors experimentally in a Reynolds number range of 3,000 to 100,000. They used an array of 19 Zircaloy rods at $p/d = 1.12$. All tests were made under isothermal conditions. It was seen that the experimental friction factors for $p/d = 1.12$ lie approximately below those for smooth tubes.

Sabbotin, Ushakov and Gabrianovich[19] have investigated the hydraulic resistance for longitudinal flow of liquids

through a square bundle of rods at p/d = 1.0 and 1.13. The bundle consisted of 7 rods and 12 spacers. Their experiment conducted with water lead to the following conclusions.

1. The experimental points for the closed packed bundle lie approximately 40 percent lower than the curves for Blasius formula. The results of the experiments for p/d = 1.13 lie 10 - 13 percent higher than these curves.

2. Less marked dependance of friction factor on Reynolds number was found for p/d = 1.13 than was found for smooth round tubes.

Experimental data are available upto p/d = 1.75. It can be concluded, in general, that both heat transfer co-efficient and friction factor increase with increased rod spacing, that both lie above the circular tube correlation for p/d 1.10, and there can be appreciable uncertainty in the magnitude of the Nusselt number and friction factor for p/d 1.10.

Although circumferential temperature variation does not appear to be prominent in arrays of p/d 1.10, it is of concern in more closely packed rod bundles at low Reynolds numbers. Evidence for the absence of interchannel mixing is inconclusive.

No attempt has been made (excepting that of Friedland et al) to study the characteristics of the bundle as a whole or

to study possible variation in rod characteristics with regard to its position in the bundle. So far, attention has been focussed on a single rod as the unit in the array. Most of the investigators simulate an infinite array with clusters of 7 rods, by providing dummy rods along the periphery. The resulting hydrodynamic picture is definitely similar to that of an infinite array, but not when a temperature field is present. This can be justified, if, in practice always large bundles of rods are used. But this is not the case, most often, in nuclear power reactors. Hence it is desirable to know the effects of the finiteness of the array. The present investigation is made with a view to find these effects.

The present work is a continuation of the work done by R.V.G. Menon. He conducted the experiments in a 19 rod delta array with p/d ratio of 1.1. He found that the heat transfer co-efficients were about 20% lower than those given by Colburn relation for circular tubes. He did not report any data about friction factor. He also found evidence for interchannel mixing.

CHAPTER III

EXPERIMENTAL SET UP

Axial flow heat transfer, temperature distribution and pressure drop studies have been made in parallel rod arrays with pitch to diameter ratios of 1.1 and 1.2. The arrays consisted of 19 rods arranged in delta geometry. A 5 1/2 inch diameter flow channel for p/d = 1.1 and a 6 inch diameter flow channel for p/d = 1.2 were used. Air was the cooling medium. Heat flux was generated in rods by electrical heating. A maximum Reynolds number of about 11,300 was realised.

Figure 4 shows a schematic diagram of the set up.

The test elements consisted of nineteen, 1 inch O.D., 0.8 inch I.D., 30 inch long, stainless steel tubes, arranged in an equilateral triangular array at centre to centre spacings of 1.1 and 1.2. 1/2 inch O.D. stainless steel tubes were connected to the ends of 1 inch O.D. stainless steel tubes (Figure 6) with the use of Brass bushes. Spacing and support were provided by precision machined end plates, with 19 properly spaced 1/2 inch diameter holes in each, to receive the free ends of the 1/2 inch O.D. tubes. Intermediate spacers were not included. The test element assemblies were fitted into a 5 1/2 inch I.D. and a 6 inch I.D., W.I.

flow channels for $p/d = 1.1$ and $p/d = 1.2$ respectively. The flow channels were closed at both ends by threaded caps.

Commercial heaters were used to internally heat the test rods (i.e., the 1 inch O.D. S.S. tubes). Each heating rod was 36 inches long, had a resistance of 16 ohms and a maximum capacity of 1 KW. The actual heated length was only 24 inches. The rods were centrally located and supported by the 1/2 inch O.D. end tubes. Additional intermediate support was provided by china clay rings. The power leads were taken out through the end tubes. Different experimental conditions were simulated through the use of control switches which were so arranged that it was possible to heat any assortment of rods at a time.

A combination of three 3/4 H.P. axial flow blowers and a 5 H.P. centrifugal compressor connected in series yielded a flow rate of about 480 lbs/hr. Another centrifugal compressor of higher capacity gave a flow rate of about 900 lbs/hr. Experiments could not be conducted at higher flow rates than these owing to lack of proper equipment. Air entered and left the test sections through 2 inch pipes normal to the direction of flow. Allowance was made for normal entry and for entrance length requirements to achieve fully developed flow conditions, by keeping the measuring section well after 20 equivalent diameters.

Measurement of temperature is one of the most tricky problems in such experiments. The test rods are not easily accessible and we cannot disturb the flow by external probes. 30 gauge

Iron - constantan thermocouples with teflon sheaths were used to measure the rod surface temperatures. These thermocouples were embedded in furrows cut longitudinally on the surface of the rods. The hot junctions were at a distance of 21 inches from the inlet end of the 1 inch S.S. tube. The junctions were placed in holes (diameter = 0.8 mm) drilled carefully at the end of the furrows (i.e. at measurement points). The junctions were then peened into these holes using a specially designed peening tool. A coating of Aluminium amalgum in Mercury was given on these holes to ensure good heat conduction. Inspite of the greatest care and precautions, it was found that the very minute differences in the nature of the bonding of the hot junctions to the rod surface caused some discrepancies in the measurements. The extent of these "initial errors" was exactly determined by suspending each of these rods, vertically, in an enclosure free of drafts or thermal currents, and heating them to the test conditions. Because of the free convection cooling, all the thermocouples should indicate the same temperature. The corrections required were calculated and later applied to test results. One thermocouple had been initially calibrated against the melting point of Zinc and the rest were compared with it. Departures from standard calibration charts were found to be negligible.

32 thermocouples were distributed on 4 test rods (Figure 8). By properly choosing the rods and the peripheral locations of the thermocouples, it was possible to get a comprehensive picture of the entire temperature field.

The thermocouple leads were taken to a cold junction, kept at 0°C by powdered ice, and then to a set of rotary selector switches. Easy and quick selection of thermocouples was accomplished by these switches. A Leeds and Northup Millivolt Potentiometer was used to measure the responses.

The air flow rate was determined by interposing an orifice-meter in the coolant circuit. The orifice was precision machined and installed as per ASME Codes and standard calibration charts were used for calculating the flow rates.

The pressure drop in a distance of 18" in the test section was also measured after allowing for the flow development, using an inclined manometer, capable of reading upto a difference of 0.01 inches of water.

CHAPTER IV

RESULTS, DISCUSSIONS AND CONCLUSIONS

The following experiments were conducted for the array with p/d = 1.1, at a flow rate of 280 lbs/hr.

<u>Test 1</u>: With the central rod (Rod 1) alone heated.

<u>Test 2</u>: With the inner ring of rods (Rods 2 to 7) heated.

<u>Test 3</u>: With all the inner rods (Rods 1 to 7) heated.

<u>Test 4</u>: With the outer ring of rods (Rods 8 to 19) heated.

<u>Test 5</u>: With all the rods heated.

<u>Tests 6 to 10</u> : Five tests were conducted with all the rods heated at flow rates of 186 lbs/hr, 350 lbs/hr, 420 lbs/hr, 685 lbs/hr and 815 lbs/hr.

The following tests were performed after changing the p/d ratio to 1.2.

<u>Tests 11 to 15</u>: With all the rods heated at flow rates of 132 lbs/hr, 270 lbs/hr, 470 lbs/hr, 780 lbs/hr and 945 lbs/hr.

Tests 1 and 4 were performed with a view to get information on interchannel mixing. Test 2 was conducted to find the effect of providing dummy rods on temperature distribution. Tests 5 to 10 were conducted to know the effect of finiteness of the rod bundle. Data on heat transfer co-efficients and friction factors was sought to be obtained by tests 5 to 10 and 11 to 15. The effect

of increased rod spacing on heat transfer co-efficients and friction factors was planned to be obtained by tests 11 to 15. Test 3 was conducted to get additional information, if any.

In each case the surface temperatures at various angular positions were measured for rods 1, 2, 10 and 11. Pressure drop in 18 inch of test section was also measured by means of an inclined manometer. The temperature distribution has been plotted in the form of 'contour' (Figures 7 to 19). It is hoped that this form of presentation will enable easy comprehension of the circumferential variation of temperature. The centres of the plots coincide with the centres of the corresponding rods. But the rods are not drawn to scale.

The flow rates were calculated (Appendix I) using the standard calibration charts for the orificemeter.

Heat transfer co-efficients were determined at different flow rates (Appendix II) corresponding to tests 7 to 10 and 11 to 15.

Friction factors were determined for various Reynolds numbers and different p/d ratios using the procedure given in Appendix III.

The results show that the Nusselt numbers obtained for p/d ratio of 1.1 are about 20 - 30% less than those given by the colburn equation for circular tube flow. For p/d = 1.2 Nusselt numbers exceed those for circular tube flow by about 5 - 10%.

This is in agreement with the pattern found by the previous investigations, i.e., for very close bundles the heat transfer rates are substantially less than those for circular tubes and for p/d 1.15 the heat transfer rates are more than those for circular tubes. This comparison can really be made only for the central rod. It is difficult to define an overall Nusselt number for the assembly, since an average wall temperature has hardly any reference to the present conditions.

For $p/d = 1.1$, the friction factors determined on the basis of the hydraulic equivalent diameter were found to be about 20 - 30% less than those given by Blasius equation for turbulent flows in circular tubes. But for $p/d = 1.2$, the friction factors are found to be greater by 15 - 20% than those for circular tubes. This is in favour of the conclusions drawn by previous investigators.

The following facts are revealed by a study of the temperature profiles.

<u>Test 4 (Figure 10)</u>: Outer rods heated, $p/d = 1.1$.

Rod 11 has a lower temperature compared to rod 10. This may be due to difference in coolant contact. The thermocouples 2 and 4 on rod 1 show a higher temperature than at other locations inspite of the fact that they are exposed to more coolant contact. This means that the heating of the rod is more due to the contact with the coolant already heated up by the peripheral rods than due to radiation from other rods. This is strong evidence for the view that interchannel mixing is not insignificant, atleast in finite

arrays. The low temperatures of rods 2 and 1 are self - explanatory.

<u>Test 2 (Figure 9):</u> Middle ring of rods heated, p/d = 1.1.

Rod 10 is cooler than rod 11. This may be explained by observing that rod 11 is in the immediate vicinity of two heated rods whereas rod 10 has only one heated rod adjacent to it. Though rod 1 is not heated it shows considerably high temperatures. This is because it is surrounded by six heated rods.

<u>Test 3 (Figure 8):</u> All the inner rods heated, p/d = 1.1.

This test shows that when dummy rods are provided, the temperature distribution will still be distorted, unless the dummy rods themselves are heated. Rod 10 is hotter than rod 11. This is self explanatory.

<u>Test 1 (Figure 7):</u> Rod 1 alone heated, p/d = 1.1

Rod 11 has peaks on those locations exposed to the air coming out of the inner channels, Where it is heated up. This is in favour of the conclusion that interchannel mixing is quite strong.

<u>Tests 5, 7, 8, 9, and 10 (Figures 11, 12, 13, 14 and 15):</u> All rods heated at different flow rates, p/d = 1.1

From these tests conducted at different Reynolds numbers, it can be observed from the temperature distributions that the effect of finiteness are manifestly evident. The temperature distributions

on rod 2 shows that the effect of wall and peripheral channels penetrate into second ring also, to a little extent. Rod 1 exhibits the expected infinite array distribution; the magnitude variation is quite small.

Tests 11 to 15 (Figures 16 to 19): All rods heated at varying flow rates, p/d = 1.2.

The relative variation in temperature distribution among rods 1, 2, 10 and 11 seem to be quite similar to those tests conducted for p/d = 1.1. Rod 2 still shows a symmetric temperature distribution thereby showing the effect of wall and peripheral channels. Rod 1 again exhibits the expected infinite array temperature distribution.

Conclusions:

The present investigation throws light on the influence of finiteness of the rod arrays on the temperature distribution; and yields data on friction factors in rod cluster geometries. It brings into focus the strong interchannel mixing which influences the heat transfer and flow characteristics in small assemblies. The wall effects were observed to penetrate slightly into the second ring of rods. Further the Nusselt numbers and friction factors lie below those given for circular tube correlation for p/d = 1.1, and above for p/d = 1.2.

The problem of unequal coolant distribution which leads to increased cooling of outer rods may be remedied by the provision of dummy rods along the periphery. This solves the flow distribution

problem. But still there is an imbalance in the heat flux distribution. In nuclear reactor fuel elements the available coolant capacity may be used to a maximum extent by enriching the peripheral rods toa greater extent, so that the heat flux in these rods is increased due to increased fission rate.

RESULTS ON HEAT TRANSFER COEFFICIENTS

P/d = 1.1 Re = 4425

Rod No.	Mean Rod Surface Temp. at x = 18" (°C)	Eq. area of surrounding channel (inch²)	Hyd. Eq. Dia. of channel (inch.)	Mass flow in the channel (lbs/hr)	$\bar{h}$ Btu/hr.ft.²°C	Calculated Nu	Nu as given by equation (1)
1	208	0.266	0.3375	10.9	4.78	8.4	13.3
2	200	0.265	0.3375	10.9	4.97	8.75	13.3
10	170	0.434	0.3680	15.5	4.37	8.36	14.7
11	170	0.664	0.0880	28.3	3.50	12.00	23.0

TABLE II

P/d = 1.1 Re = 5300

1	166	0.265	0.3375	13.6	6.24	11.00	15.5
2	166	0.265	0.3375	13.6	6.23	10.96	15.5
10	140	0.434	0.3680	22.2	5.62	11.30	16.5
11	140	0.664	0.6880	34.0	4.80	15.70	20.5

RESULTS ON HEAT TRANSFER COEFFICIENTS

P/d = 1.1 Re = 8250

Rod No.	Mean Rod Surface Temp. at x = 18" (°C)	Eq. area of surrounding channel ($inch^2$)	Hyd. Eq. Dia. of channel (inch)	Mass flow in the channel (lbs/hr)	$\bar{h}$ Btu/hr.ft.^{2} °C	Calculated Nu	Nu as given by equation (1)
1	97	0.265	0.3375	21.8	10.6	18.50	22.5
2	96	0.265	0.3375	21.4	10.4	18.00	22.5
10	87	0.434	0.3680	31.4	10.1	19.30	24.18
11	86	0.604	0.6880	54.0	9.06	30.80	37.95

TABLE IV

P/d = 1.1 Re = 10,800

1	96.0	0.265	0.3375	27.0	10.9	19.2	26.0
2	90.0	0.265	0.3375	27.0	10.9	19.2	26.8
10	81.0	0.434	0.3680	30.4	10.5	22.4	30.2
11	77.5	0.684	0.6880	67.0	9.6	36.8	45.2

TABLE V

RESULTS ON HEAT TRANSFER COEFFICIENTS

P/d=1.2 Re = 3300

Rod No.	Mean Rod Surface Temp. at x = 18" (°C)	Eq. area of surrounding channel (inch²)	Hyd. Eq. Dia. of channel (inch)	Mass flow in the channel (lbs/hr)	$\bar{h}$ Btu/hr.ft.² °C	Calculated Nu	Nu as given by equation (1)
1	121	0.485	0.503	9.3	3.34	12.00	10.6
2	121	0.485	0.503	9.3	3.30	12.10	10.6
10	126	0.715	0.566	14.4	3.14	10.07	9.32
11	[illegible]	0.944	0.906	19.6	3.15	16.08	15.0

TABLE VI

P/d=1.2 Re = 5050

Rod No.	Mean Rod Surface Temp. at x = 18" (°C)	Eq. area of surrounding channel (inch²)	Hyd. Eq. Dia. of channel (inch)	Mass flow in the channel (lbs/hr)	$\bar{h}$ Btu/hr.ft.² °C	Calculated Nu	Nu as given by equation (1)
1	111	0.485	0.503	16.5	6.15	18.93	17.69
2	114	0.485	0.503	16.5	6.15	18.93	17.69
10	100	0.715	0.566	25.0	6.14	18.91	17.24
11	107.5	0.944	0.906	33.3	4.73	22.34	22.00

TABLE VII

P/d = 1.2 Re = 9150

RESULTS ON HEAT TRANSFER COEFFICIENTS

Rod No.	Mean Rod Surface Temp at x = 18" (°C)	Eq. area of surrounding channel (inch²)	Hyd. Eq. Dia. of channel (inch.)	Mass flow in the channel (lbs/hr)	$\bar{h}$ Btu/hr.ft.^{2} °C	Calculated Nu	Nu as given by equation (1)
1	75	0.465	0.503	26.6	8.90	27.4	25.8
2	77	0.465	0.503	26.6	8.80	27.0	25.8
10	63	0.715	0.566	46.7	8.95	28.5	24.35
11	65	0.944	0.906	68.5	9.00	42.65	37.20

TABLE VIII

P/d = 1.2 Re = 11,300

Rod No.	Mean Rod Surface Temp at x = 18" (°C)	Eq. area of surrounding channel (inch²)	Hyd. Eq. Dia. of channel (inch.)	Mass flow in the channel (lbs/hr)	$\bar{h}$ Btu/hr.ft.^{2} °C	Calculated Nu	Nu as given by equation (1)
1	65.5	0.465	0.503	33.0	10.8	33.4	30.89
2	66.0	0.465	0.503	33.0	10.66	32.75	30.89
10.	57.0	0.715	0.566	50.0	11.3	33.6	30.90
11	58.6	0.944	0.905	68.5	10.2	43.2	43.80

RESULTS ON FRICTION FACTOR

P/d = 1.1

Reynolds Number	Pressure Drop in 18" of test section (inches of water)	Friction factor calculated by equation No. (5)	Friction factor for eq. circular tube calculated by Blasius eqn. represented by (6)
3180	0.150	0.0071	0.00948
5350	0.310	0.0062	0.00839
8280	0.710	0.0059	0.00741
10500	0.990	0.0054	0.00707

TABLE X

P/d = 1.2

3000	0.25	0.0125	0.0106
5000	0.45	0.0100	0.0081
9150	1.10	0.0090	0.0078
11800	1.60	0.0082	0.0076

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APPENDIX I

CALCULATION OF AIR FLOW RATE USING ORIFICEMETER

Data for Orificemeter 1:

Orifice diameter : D_2 = 1.4 inch.

Diameter ratio : β = 0.7

Data for Orificemeter 2:

Orifice diameter : D_2 = 2.72 inch.

Diameter ratio : β = 0.68

Air inlet temperature, t_1 = 37°C

Properties of Air at inlet temperature:

$\gamma = 1.4$, $\rho = 0.072$ lbs/cft.

The equation for flow rate used here and derived in reference (21) is :

$$W = 360.1\,(D_2)^2\,K\,\sqrt{\rho_1 h_w} \qquad (2)$$

Where,

W = flow rate in lbs/hr.

D_2 = Orifice diameter in inches

K = The Orifice discharge constant

ρ_1 = Density of air at inlet in lbs/cft

h_w = Pressure difference in inches of water for a given h_w, the only unknown is the discharge constant K. So W is in terms of K.

Using this W, the Reynolds number for flow through the Orifice is determined, in terms of K

$$Re = \frac{\rho V D_2}{\mu} = \frac{6.316}{D_2 Z} W$$

Where Z is the absolute viscosity of the fluid, in centipoises at flow temperature and pressure.

This gives a Reynolds number of flow in terms of K. We get the following two equations involving K:

$$W = N_1 K \qquad (2a)$$

and $$Re = N_2 K \qquad (2b)$$

A rough estimate of K is then made, and an approximate value of Reynolds number is determined using Equation (2a). By continued reference to the tabulated discharge coefficients for the given pipe size and diameter ratio, K is then corrected to the desired accuracy by successive approximations. The actual flow rate is then computed by substituting this value in Equation (2).

Two Orificemeters were used to determine the flow rates. One of them was an integral part of the centrifugal compressor yielding higher flow rates. Another Orificemeter was necessary for determining the flow rate when blower assembly was used.

APPENDIX II

CALCULATION OF FILM HEAT TRANSFER COEFFICIENTS

Heat transfer rate $= \dfrac{E^2}{R_{res}} = \dfrac{230}{6} \times \dfrac{230}{6 \times 15} = 98.0$ Watts/Rod

$= 338.0$ Btu/hr

Since the heater portion is only 2 feet long, heat transfer area $= \pi \times \dfrac{1}{12} \times 2 = 0.524$ sq. ft.

$\therefore$ Heat Transfer rate per unit area = 646.0 Btu/hr/sq. ft.

$\bar{q}$ (Average heat transfer rate, Btu/hr) $= \bar{h} A \Delta T$

$$\Delta T = t_{s,x} - \left(t_{m,in} - \frac{X}{x} \frac{\bar{q}}{m\, C_p} \right);$$

$t_{s,x}$ is rod surface temperature at distance

x from the beginning of the heater portion.

X is total heated length

$t_{m,in}$ is mixed mean inlet temperature.

m is mass flow in the surrounding channel

C_p is specific heat of air at inlet temperature

Therefore the average film heat transfer coefficient $\bar{h}$ is defined as:

$$\bar{h} = \frac{\bar{q}}{A \left(t_{s,x} - \dfrac{x}{X} \dfrac{\bar{q}}{m\, C_p} - t_{m,in} \right)} \qquad (4)$$

and the Nusselt Modulus

$$Nu = \frac{\bar{h}\, d_e}{K}; \text{ Where}$$

d_e is the hydraulic equivalent diameter of the surrounding channel.

The area and hydraulic equivalent diameter of surrounding channels are calculated taking dividing lines equidistant from nearest bounding walls. The mass flow in each channel is computed on a proportionate basis.

Such calculations were made for tests 7 to 10 and 11 to 15. The results have been tabulated in Tables I to VIII.

APPENDIX III

DETERMINATION OF FRICTION FACTOR

Pressure drop between two sections located at 12 inch and 30 inch from the inlet has been measured. Let the sections be denoted by 1 and 2 respectively. Since there is considerable rise in temperature of the fluid between these two sections, the flow cannot be considered to be isothermal. The pressure drop in the case of non-isothermal flow of a gas through a rod bundle is approximately given by the flowing expression (Ref. No. 20).

$$\Delta P = \frac{(\rho V)^2}{\rho_m g} \left(\frac{2 fL}{D_{eq}} + \frac{\Delta T}{T_m} + \ln \frac{P_i}{P_o} \right) \qquad (5)$$

Where,

$$P_i = \frac{2P_m + \Delta P}{2}, \quad P_o = \frac{2P_m - \Delta P}{2}$$

P_m = mean coolant temperature $= \rho_m R T_m$

$$\Delta T = T_2 - T_1, \quad T_m = \frac{T_1 + T_2}{2}$$

Where,

T_1 and T_2 are mean bulk temperatures at sections 1 and 2 respectively.

$$\rho_m = \frac{\rho_1 T_1 + \rho_2 T_2}{T_1 + T_2}$$

L = Length between two test sections

D_{eq} = Hydraulic equivalent diameter for the rod bundle

ρV = $\dot{M}/A$ by continuity equation where $\dot{M}$ is the mass flow rate and A is the flow area.

$$\rho_1 = P_1 / R T_1 \quad \text{and} \quad \rho_2 = P_2 / R T_2$$

Where R is the gas constant.

Since $\Delta P \ll P_m$, $P_1 \simeq P_2$ and hence the term $\ln (P_1/P_2)$ can be neglected in comparison with other terms. f is the only unknown and it can be determined. The results for various Reynolds numbers and different p/d ratios have been tabulated, (Refer Tables IX and X). These results are compared with those of circular tubes given by Blasius formula as follows:

$$f = 0.0791 / (Re)^{0.25} \qquad \text{(e)}$$

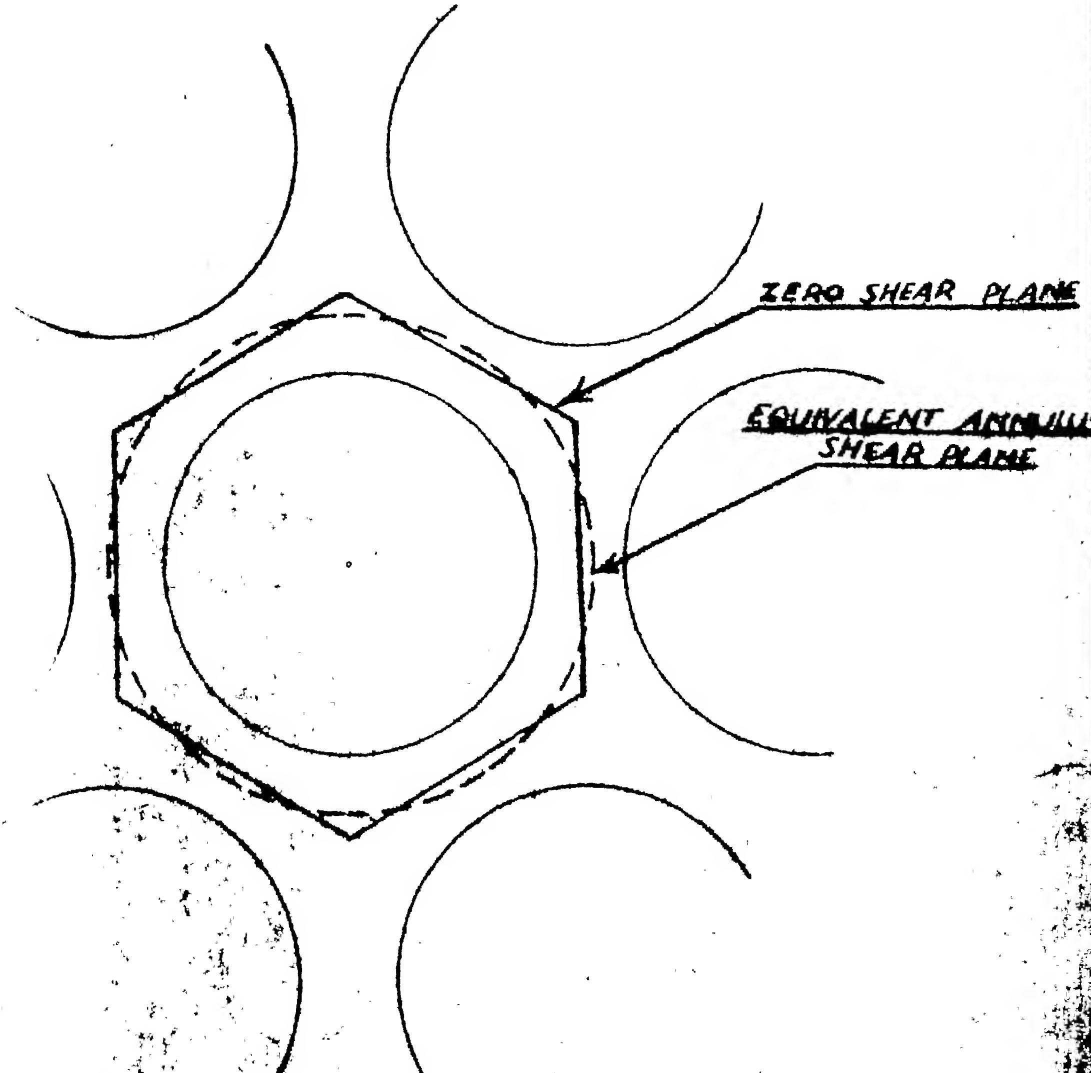


Fig . ZERO SHEAR PLANE
EQUIVALENT ANNULUS MODEL
(DWYER ET AL 1960)

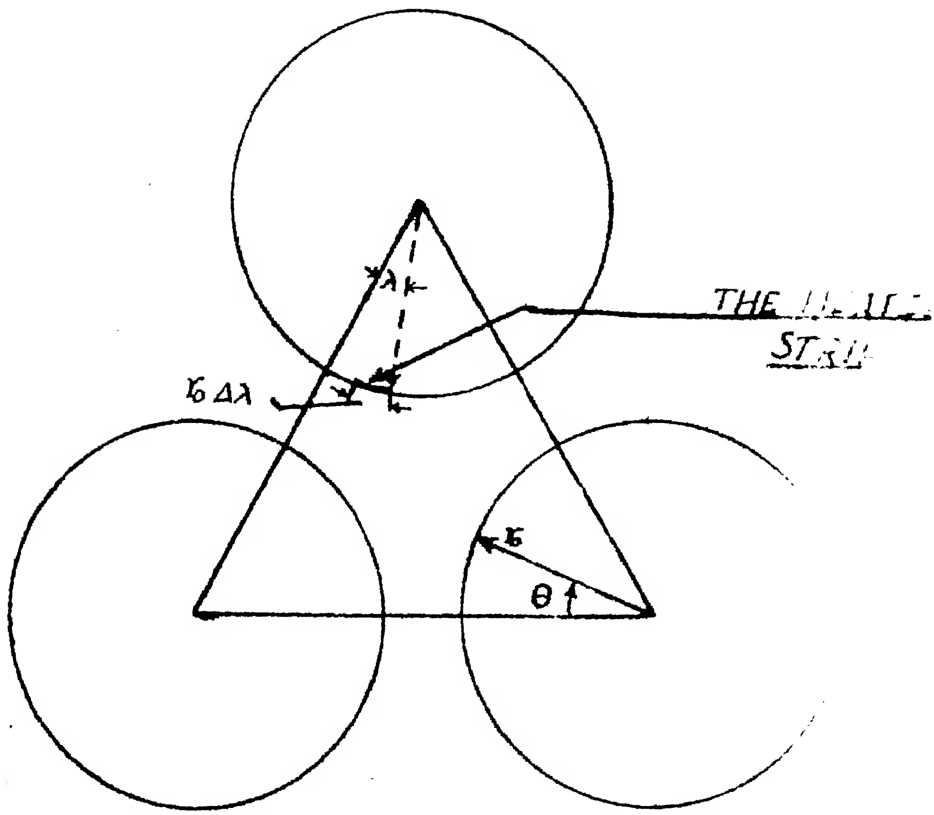


Fig 2. THE COORDINATE SYSTEM FOR THE PRINCIPAL SOLUTIONS (SUTHERLAND & KAYS 1966)

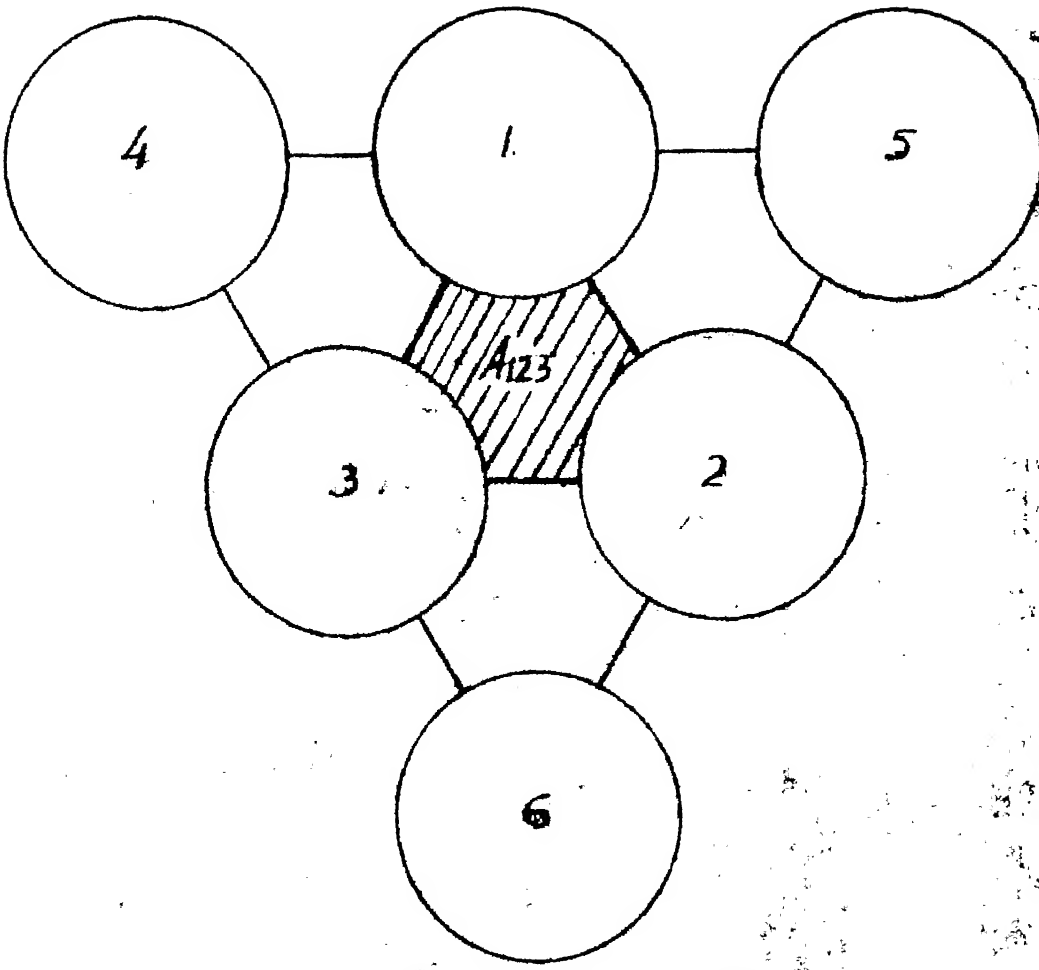


FIG 3 A UNIT FLOW AREA IN AN INFINITE ARRAY

(SUTHERLAND & KAYS 1966)

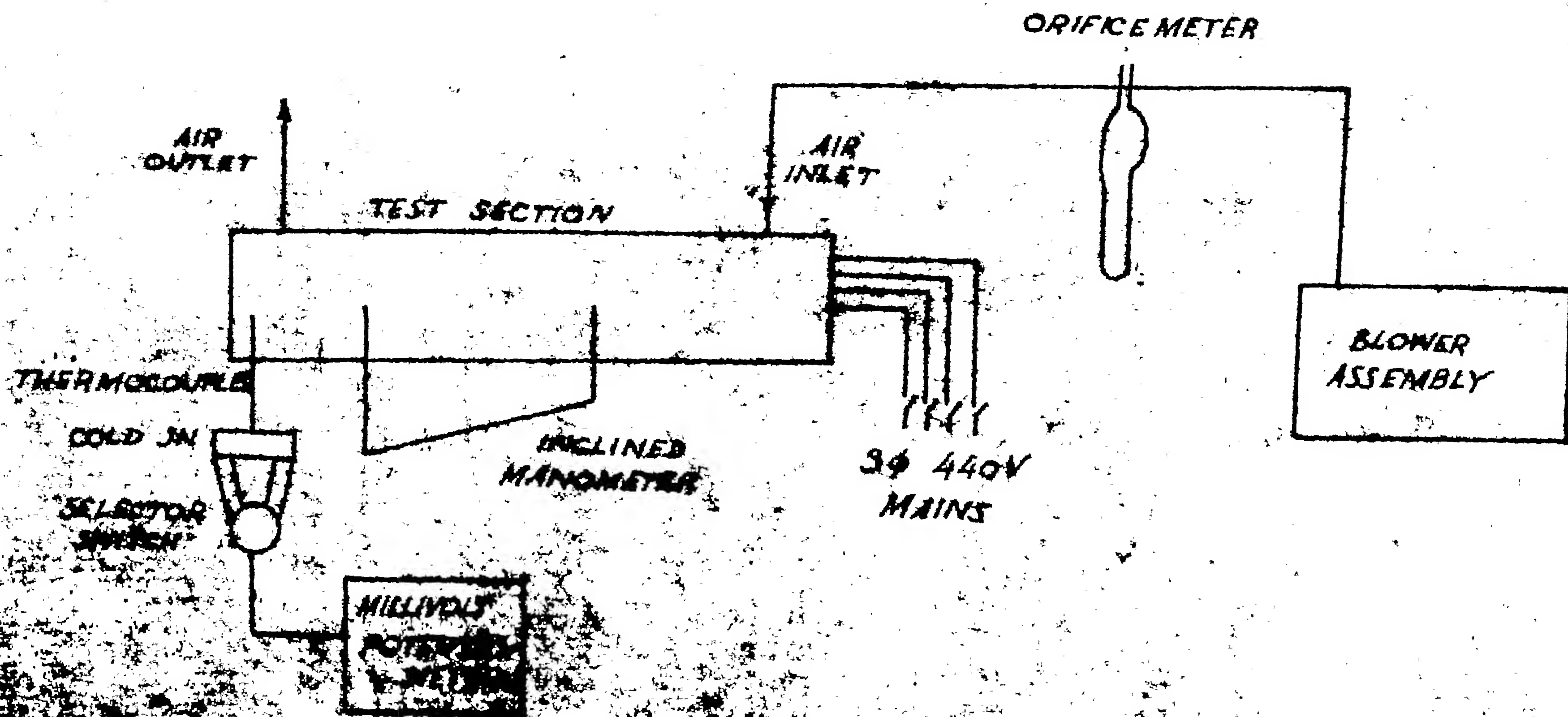


Fig 4 SCHEMATIC REPRESENTATION OF THE EXPERIMENTAL SET-UP

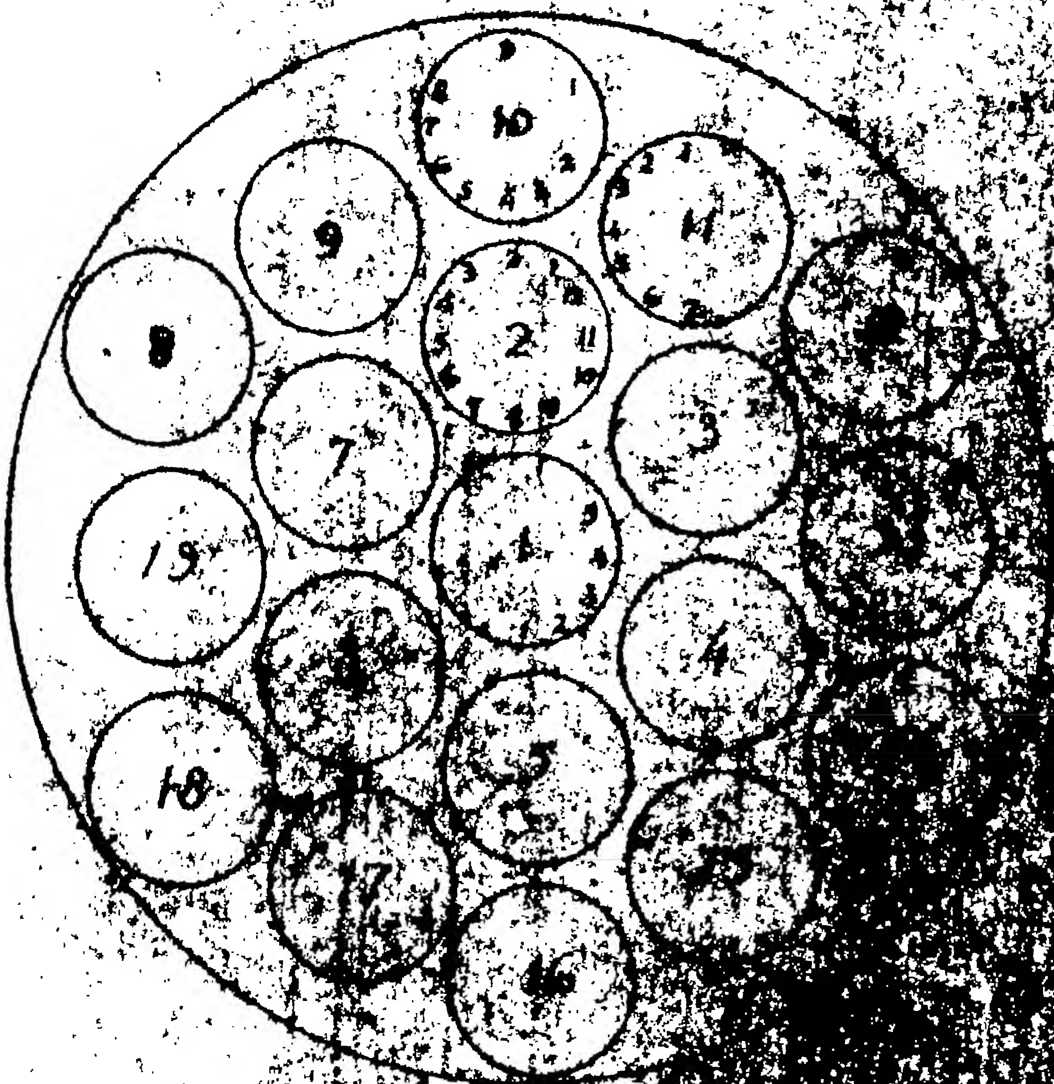


FIG. 5 ROD ARRANGEMENT

FIG. 6 TEST ROD ASSEMBLY DETAILS

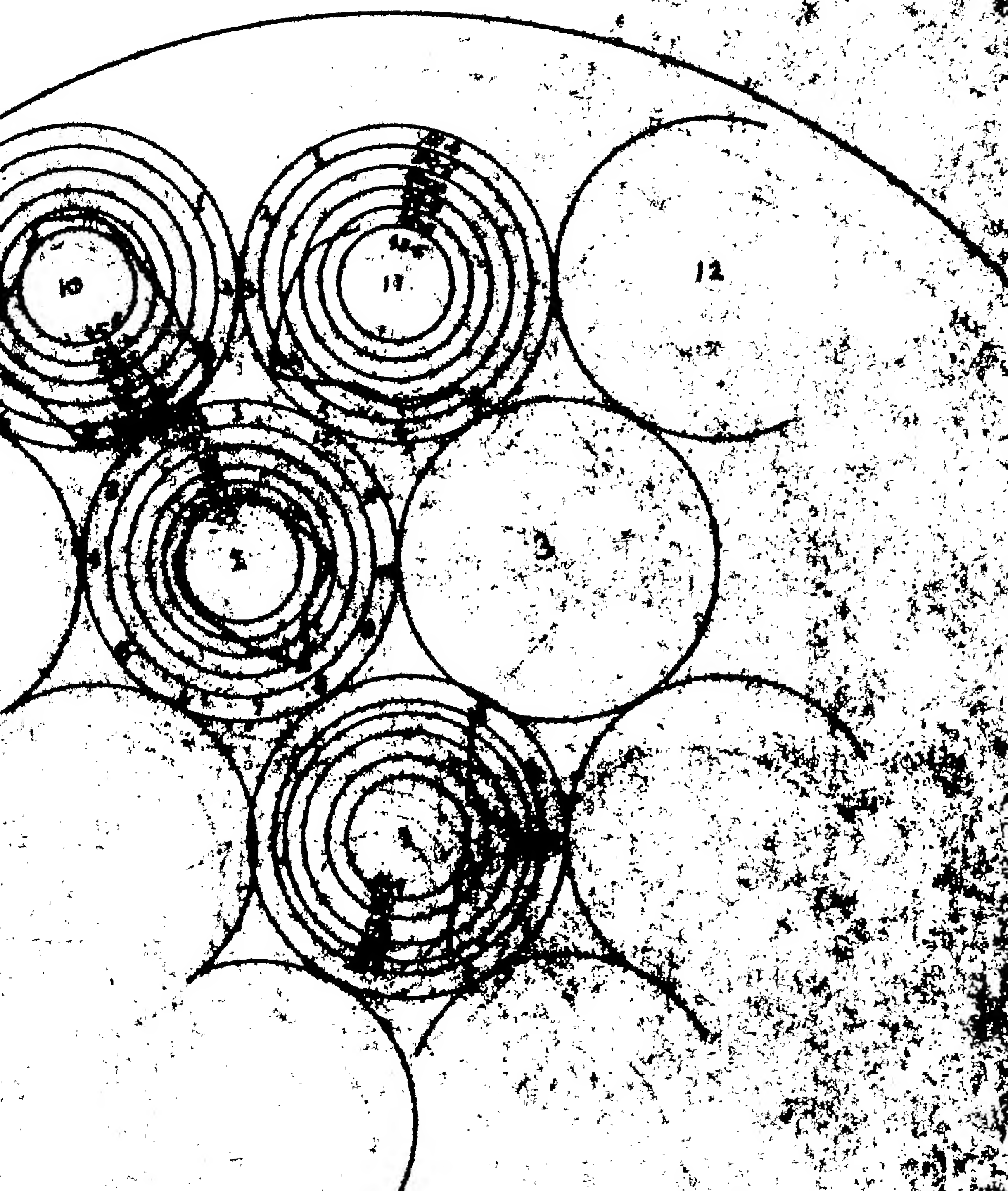


FIG 7 TEMPERATURE DISTRIBUTION
[illegible]
$P/d = 1{\cdot}1$

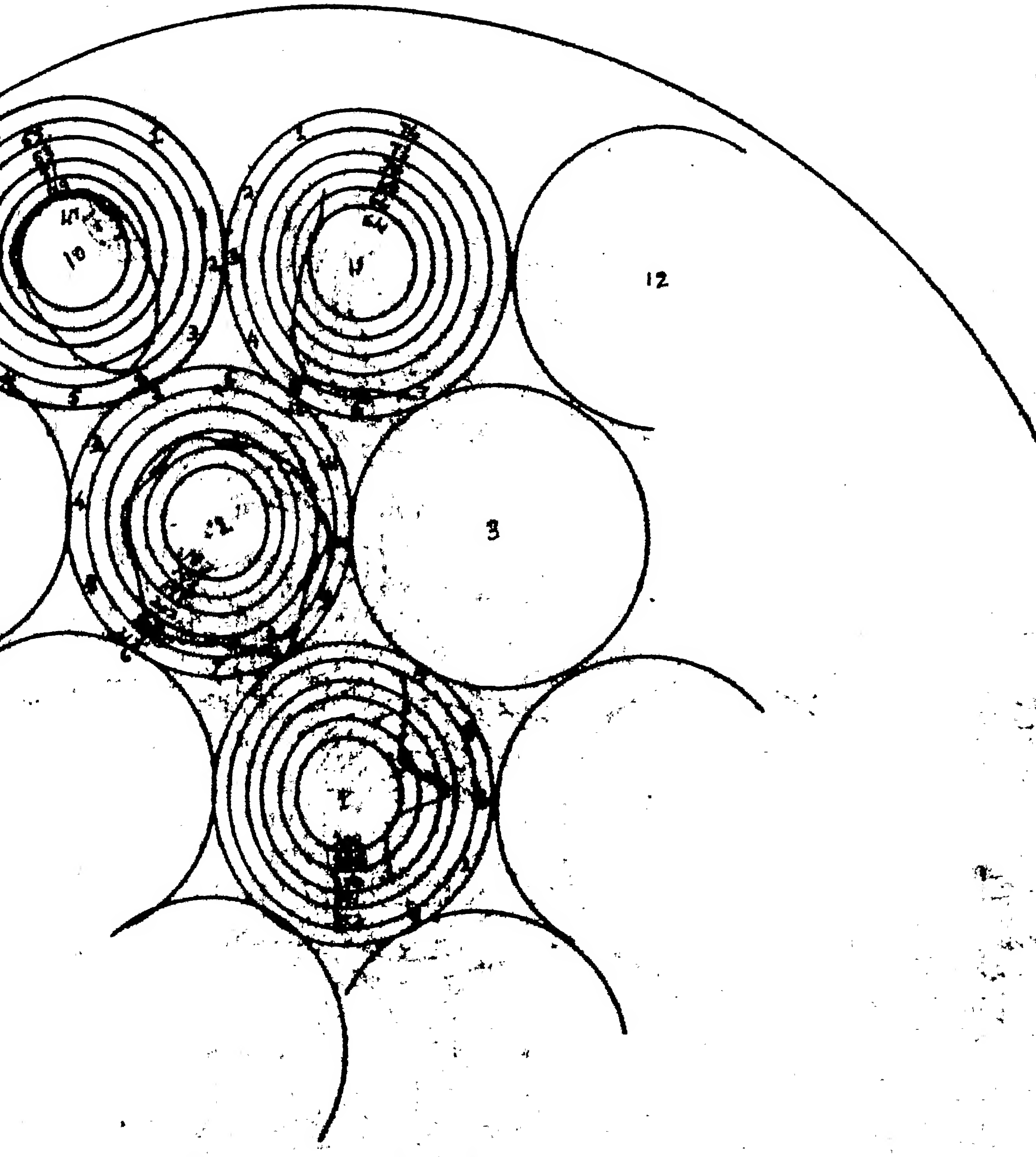


FIG. 8 TEMPERATURE DISTRIBUTION

[illegible]

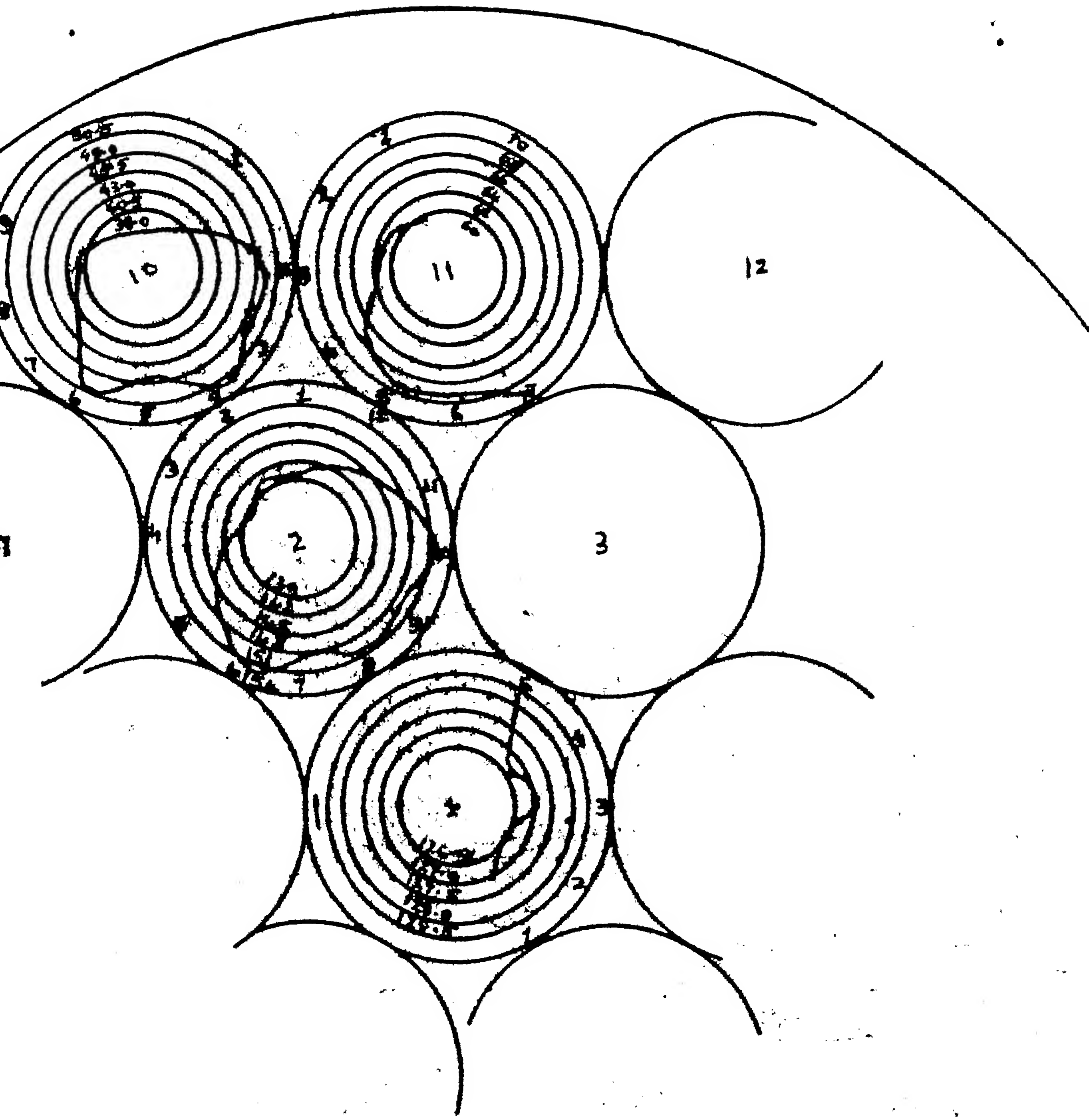


FIG. 9 TEMPERATURE DISTRIBUTION

RODS #2 TO #7 HEATED

$P/d = 1.1$

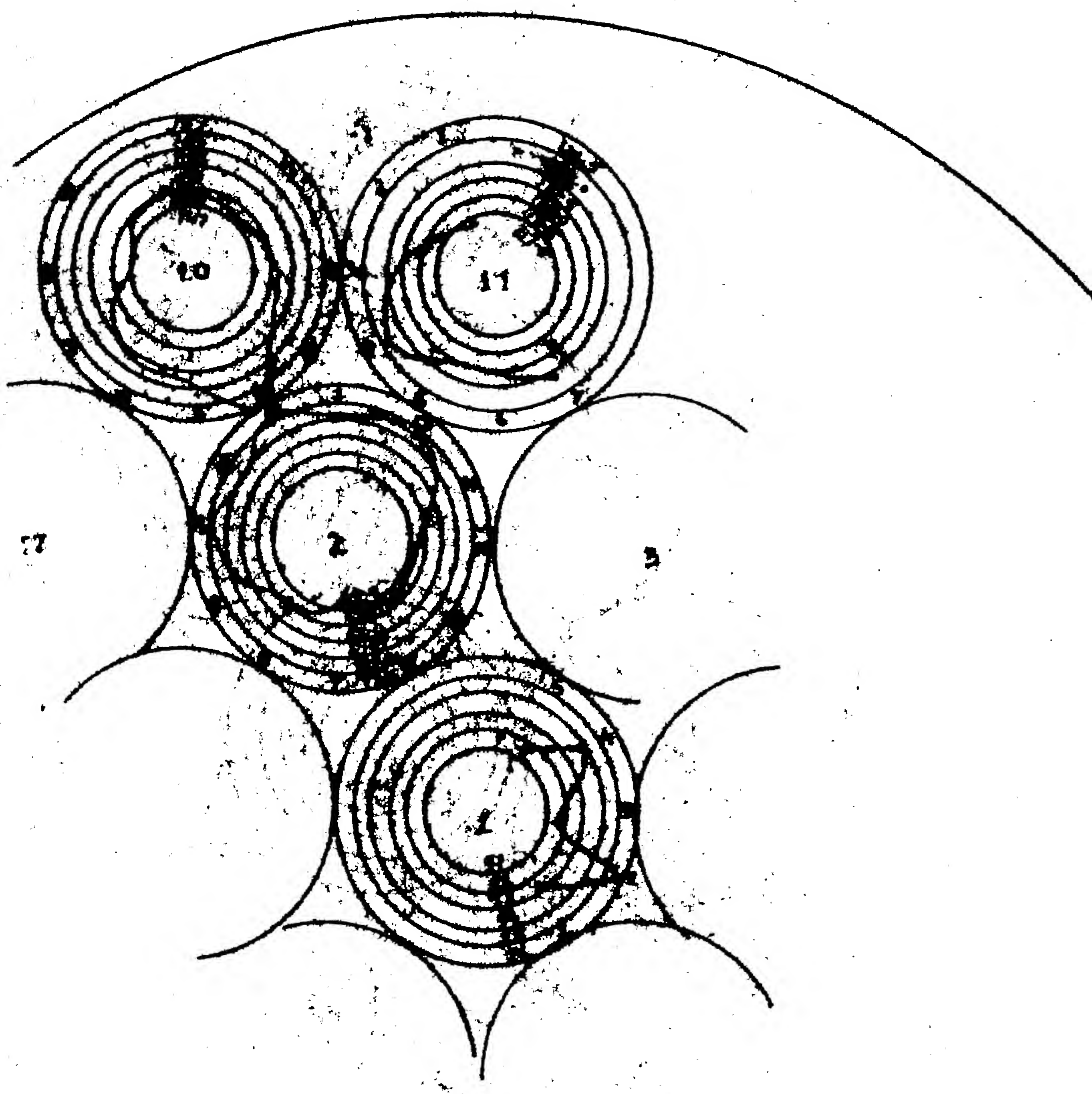


FIG 10 TEMPERATURE DISTRIBUTION

OUTER RODS [illegible] ATED

$P/d = 1.1$

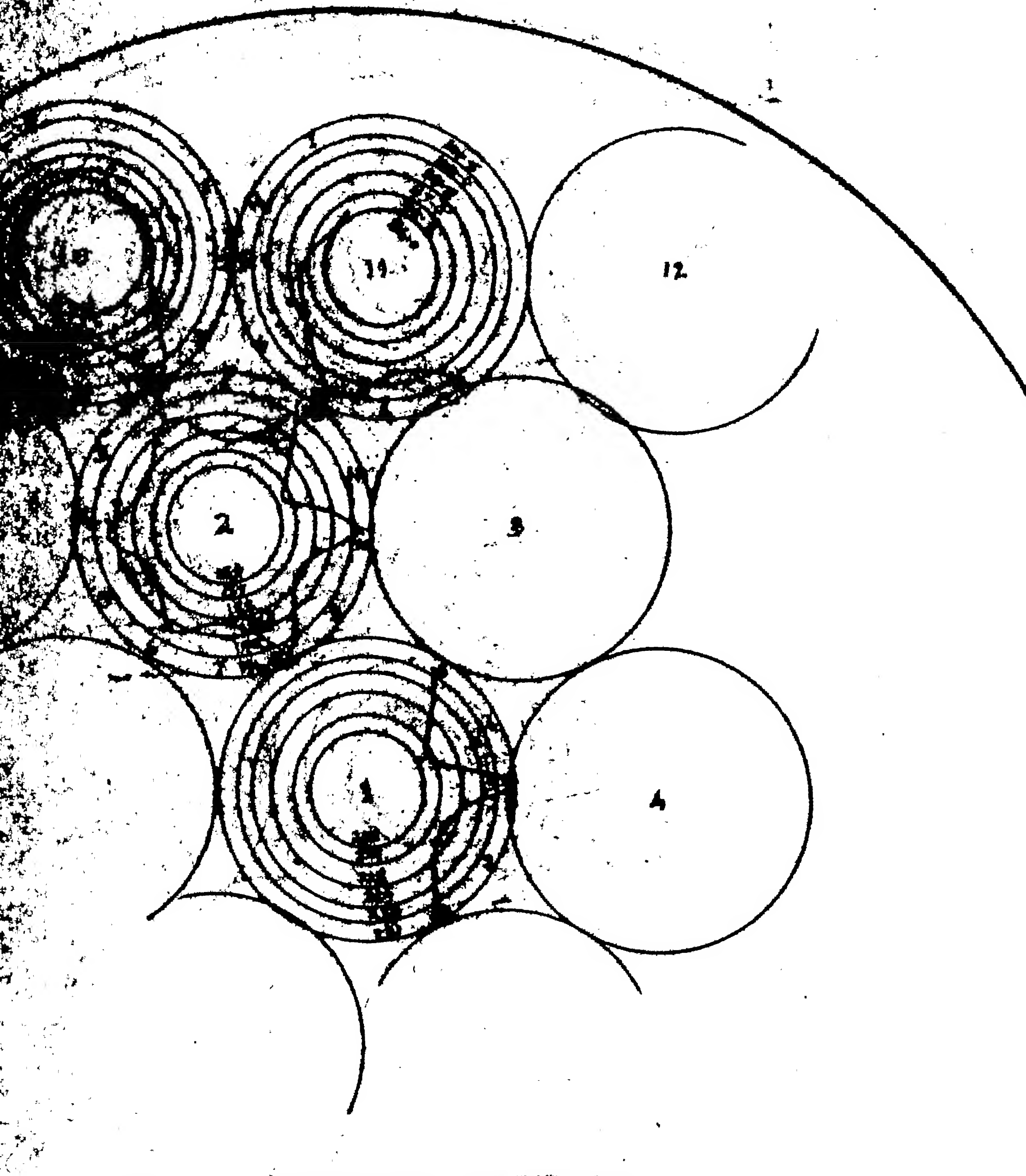


FIG 11 TEMPERATURE DISTRIBUTION

ALL RODS HEATED (#1 TO #19)

P/d = 1.1 Re = 8180

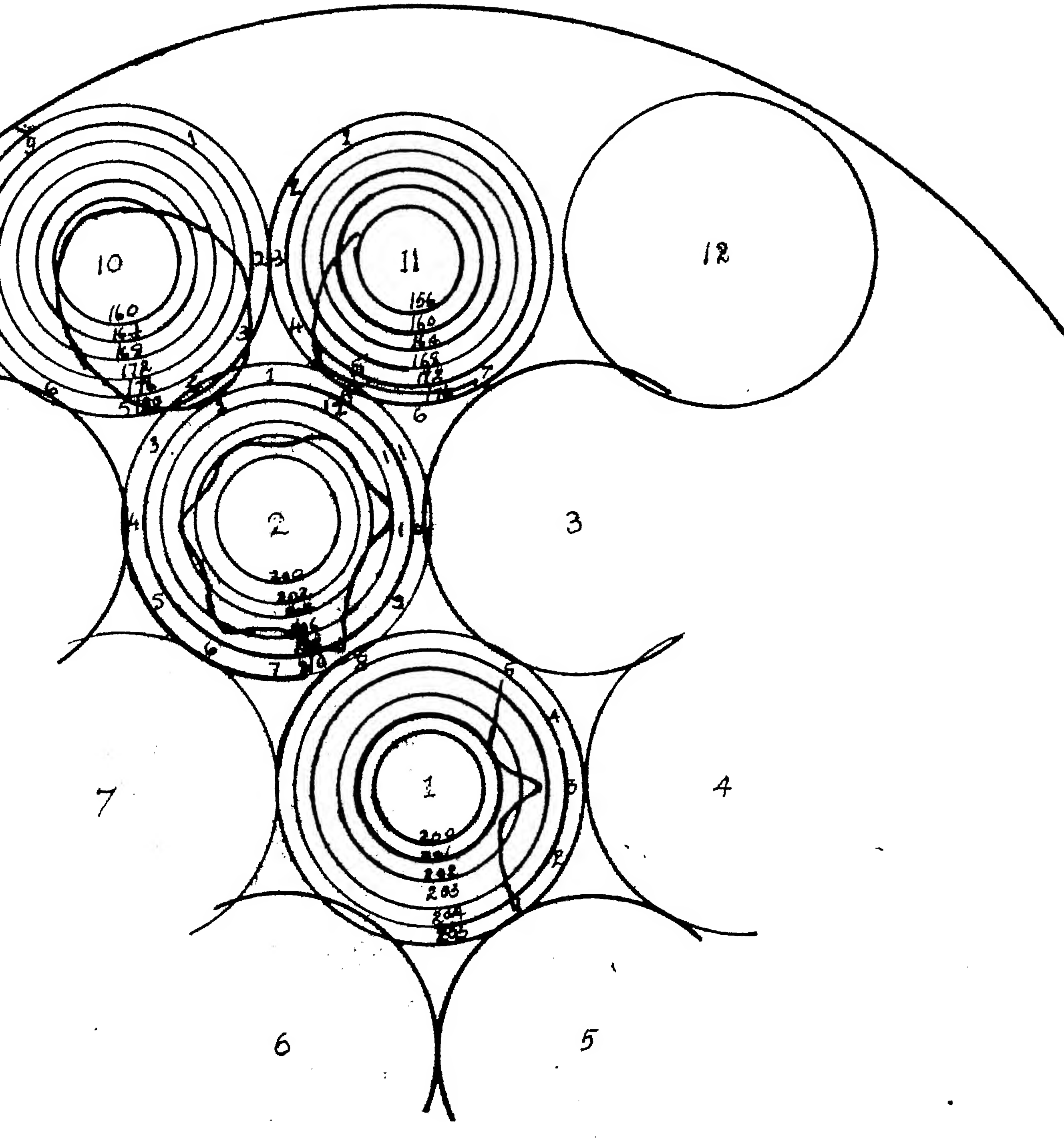


FIG 12 TEMPERATURE DISTRIBUTION
ALL RODS HEATED (# 1 TO # 19)
P/d = 1.1 Re = 4,425

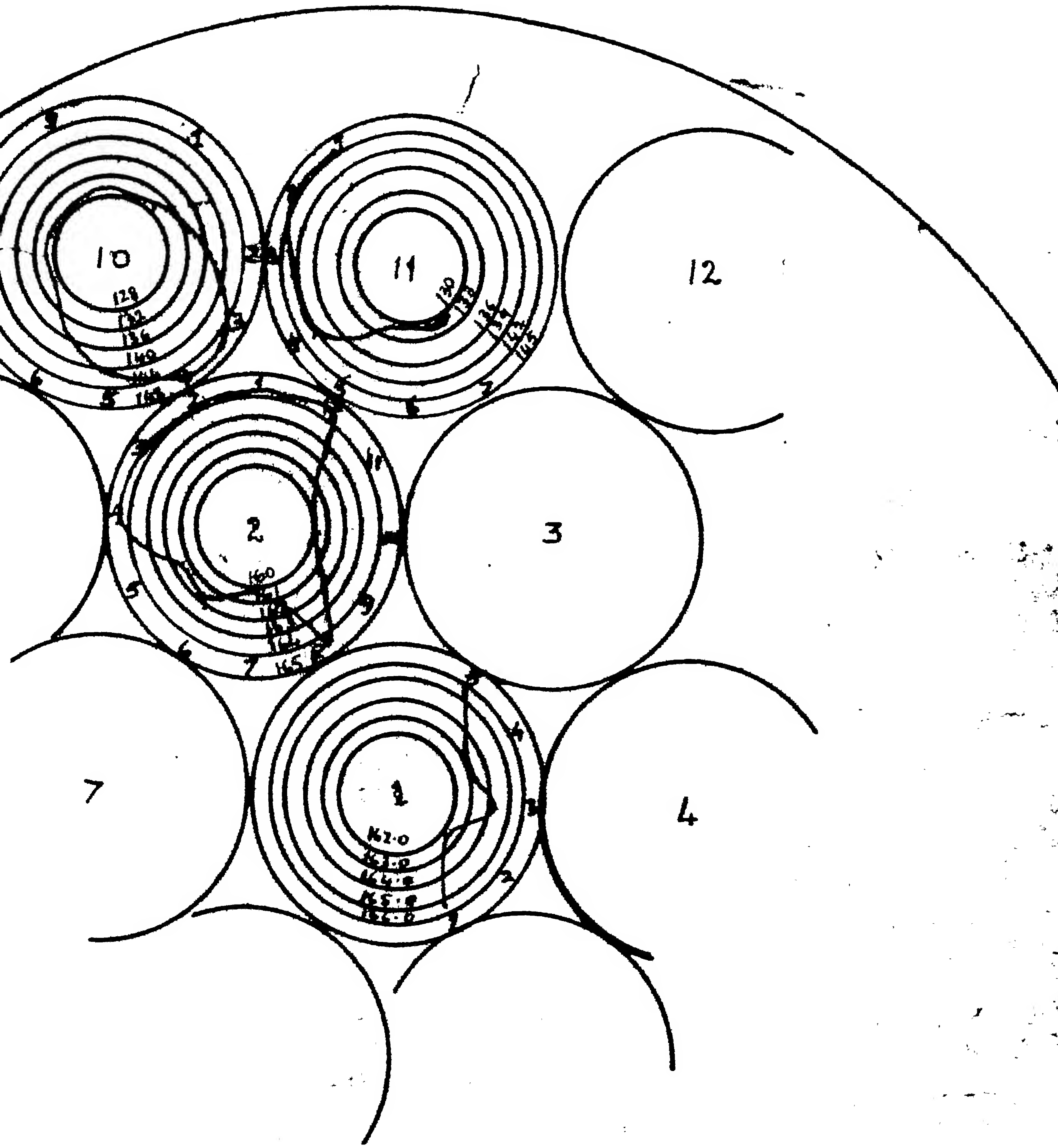


FIG 13 TEMPERATURE DISTRIBUTION

ALL RODS HEATED (# 1 TO 19)

P/d = 1.1 Re = 5250

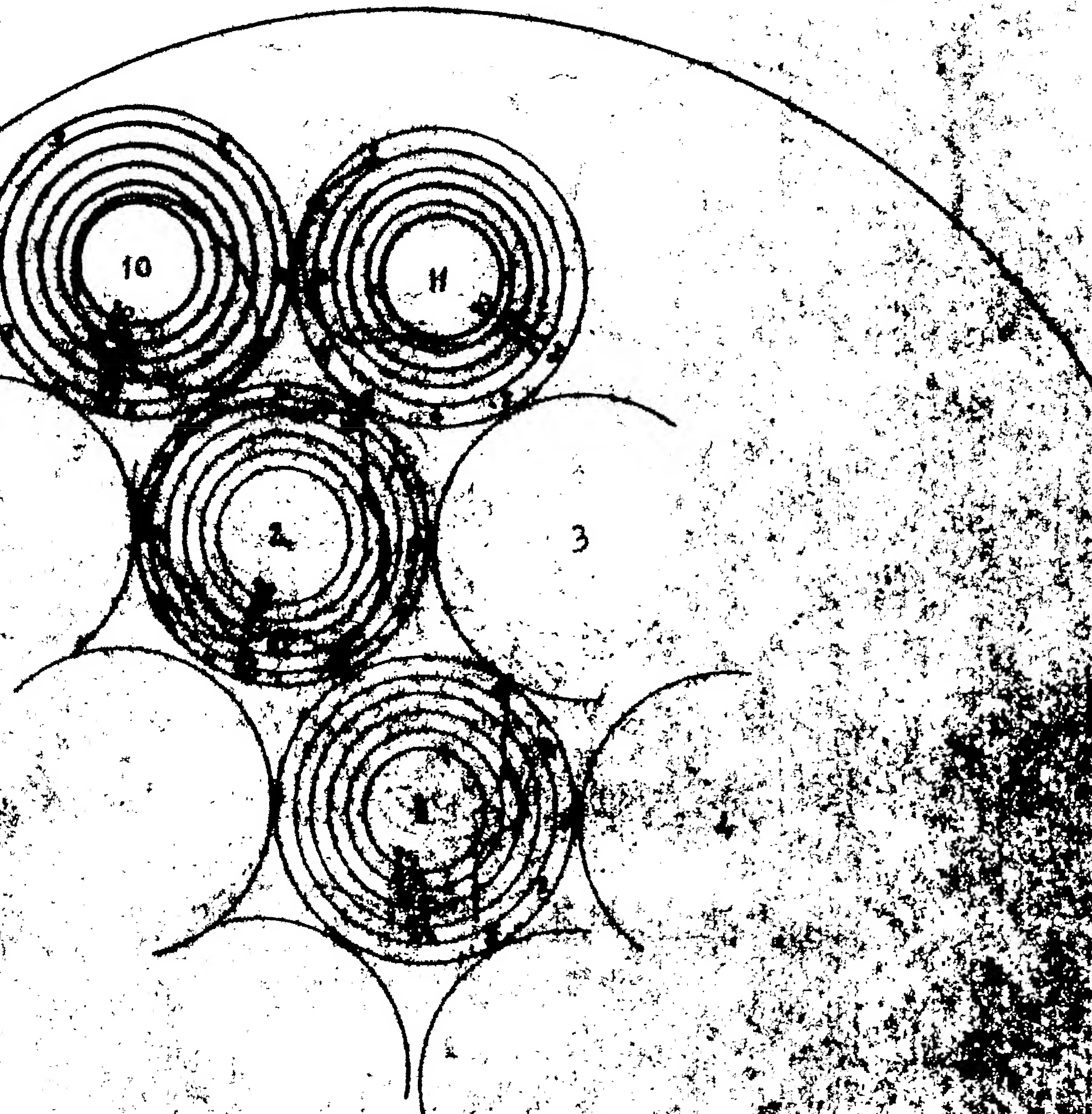


FIG. 14 TEMPERATURE DISTRIBUTION

ALL [illegible] HEATED ([illegible])

P/d = [illegible] [illegible] = [illegible]

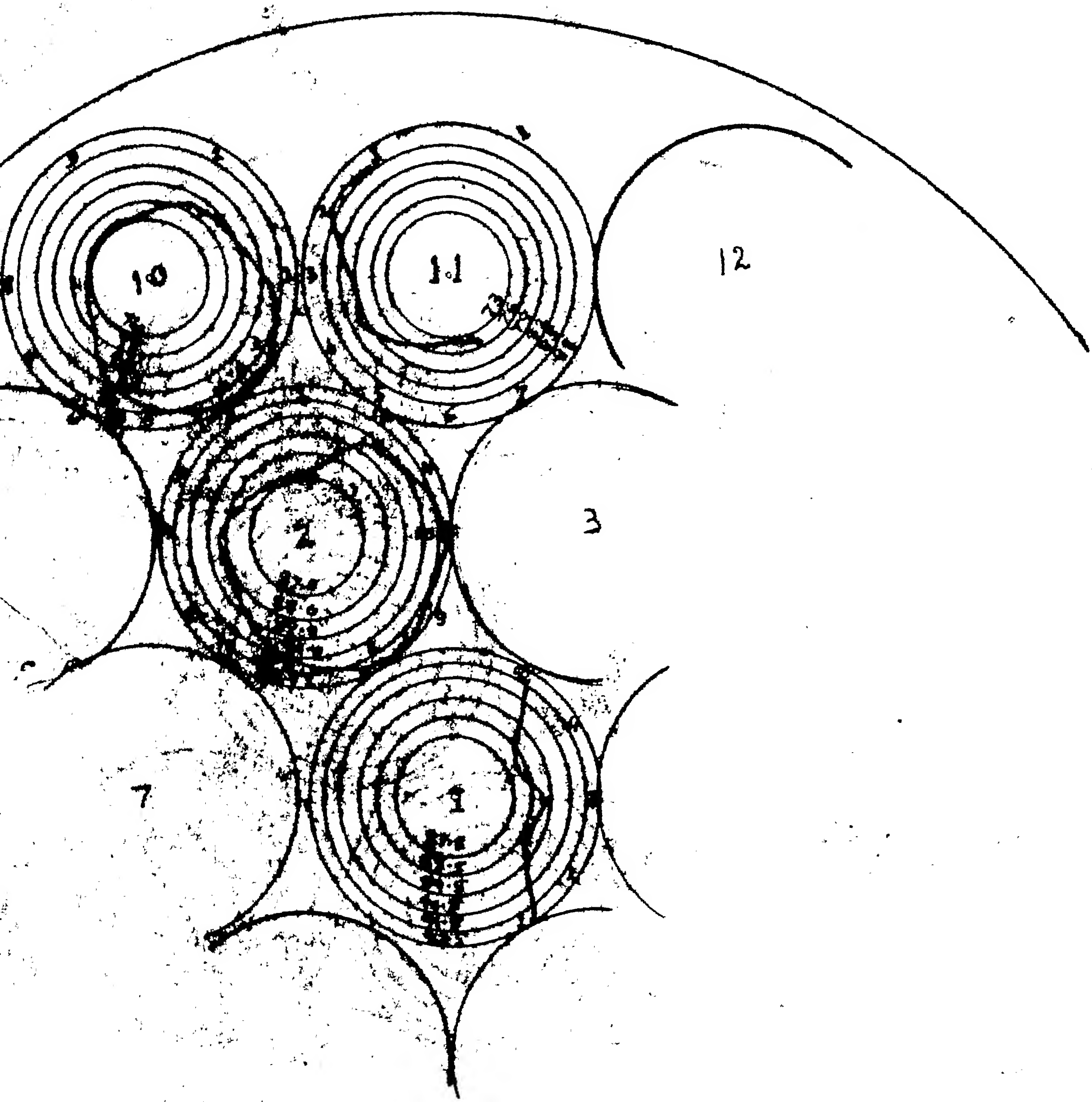


FIG 16 TEMPERATURE DISTRIBUTION
ALL RODS HEATED (# 1 TO # 19)
P/d = 1.1 Re = 10,500

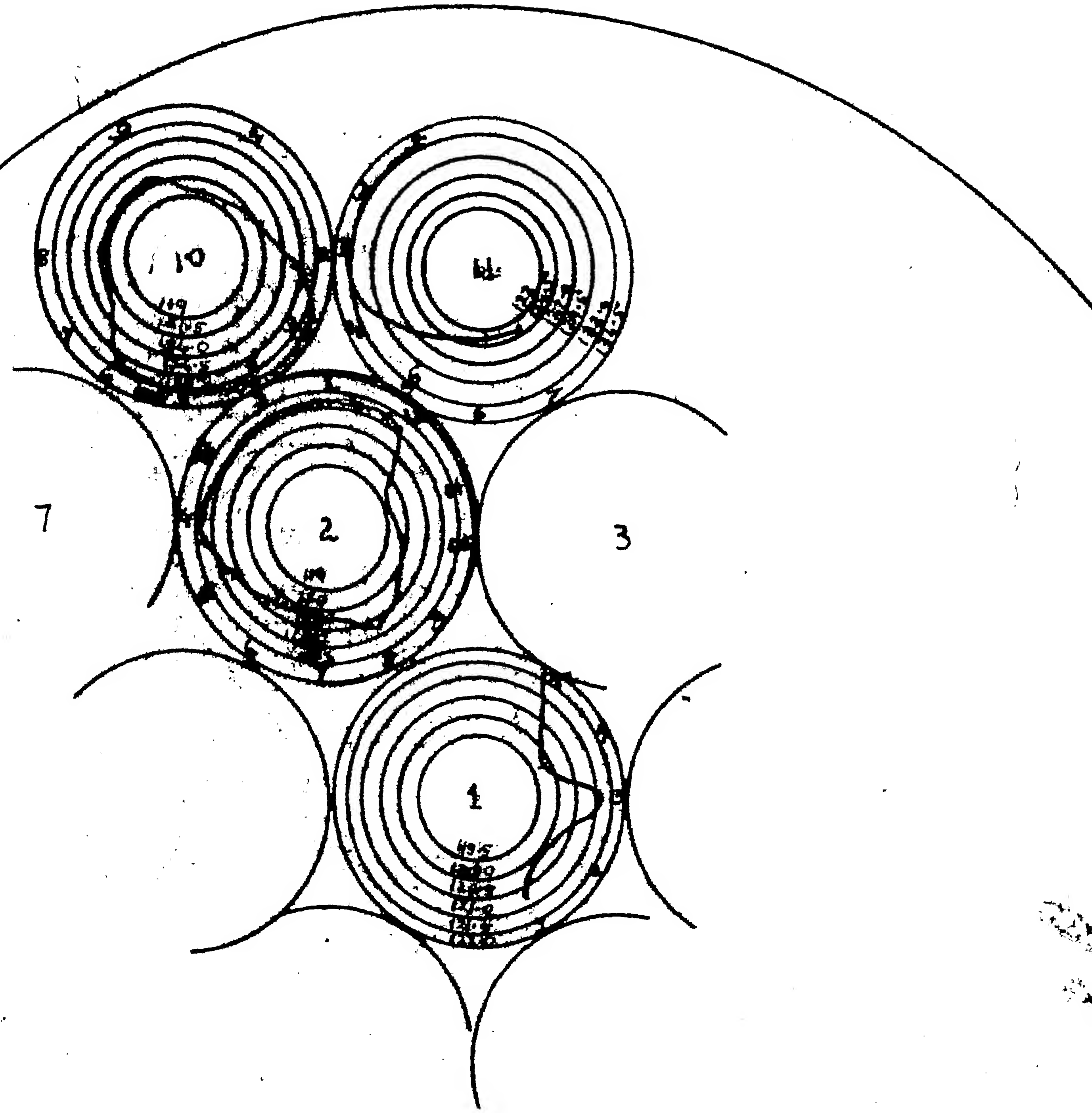


FIG 16 TEMPERATURE DISTRIBUTION

ALL RODS HEATED

P/d = 1.2 Re = 3300

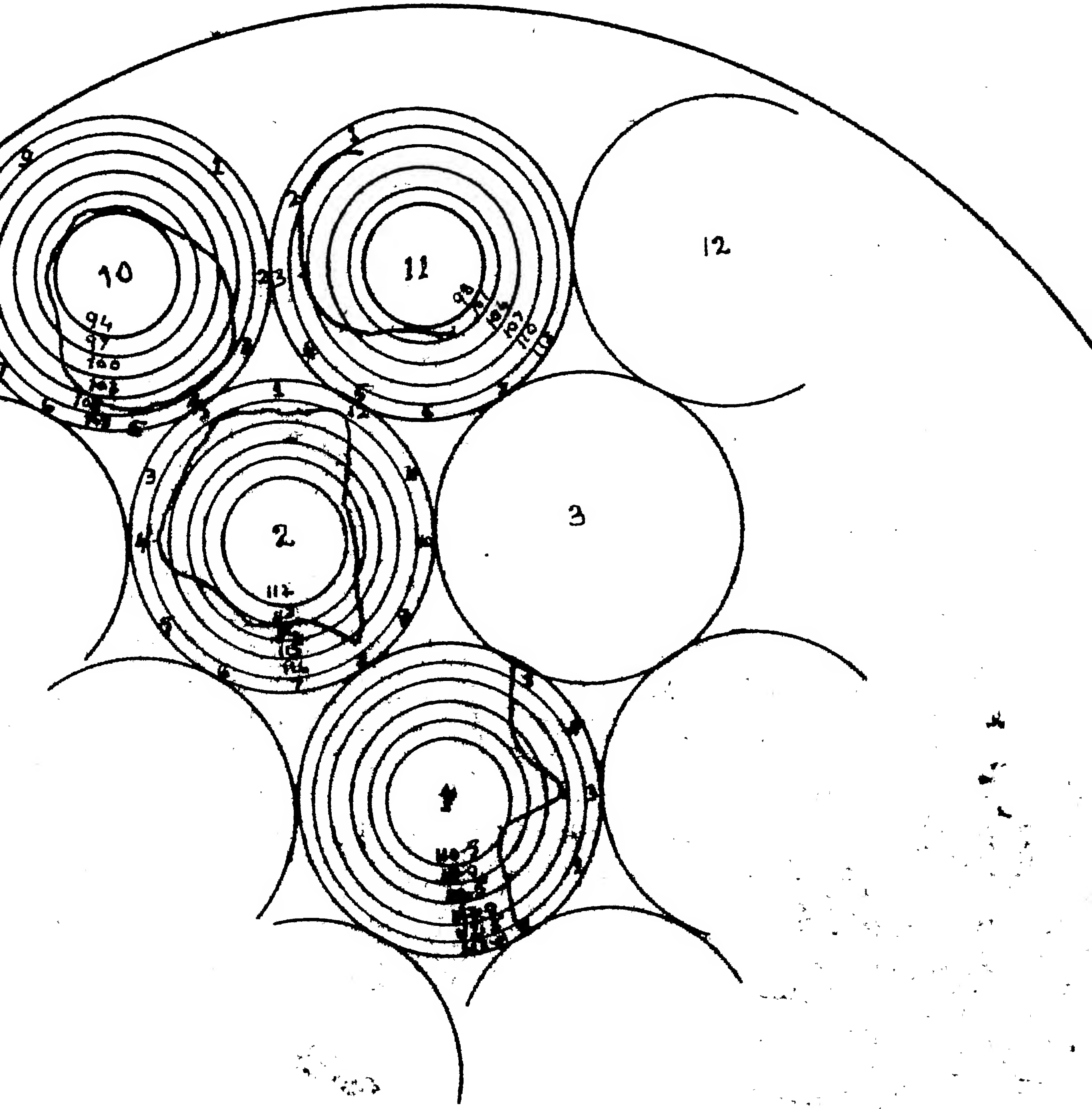


FIG 17 TEMPERATURE DISTRIBUTION

ALL RODS HEATED (#1 TO #19)

98 = 1.2 • Re = 5850

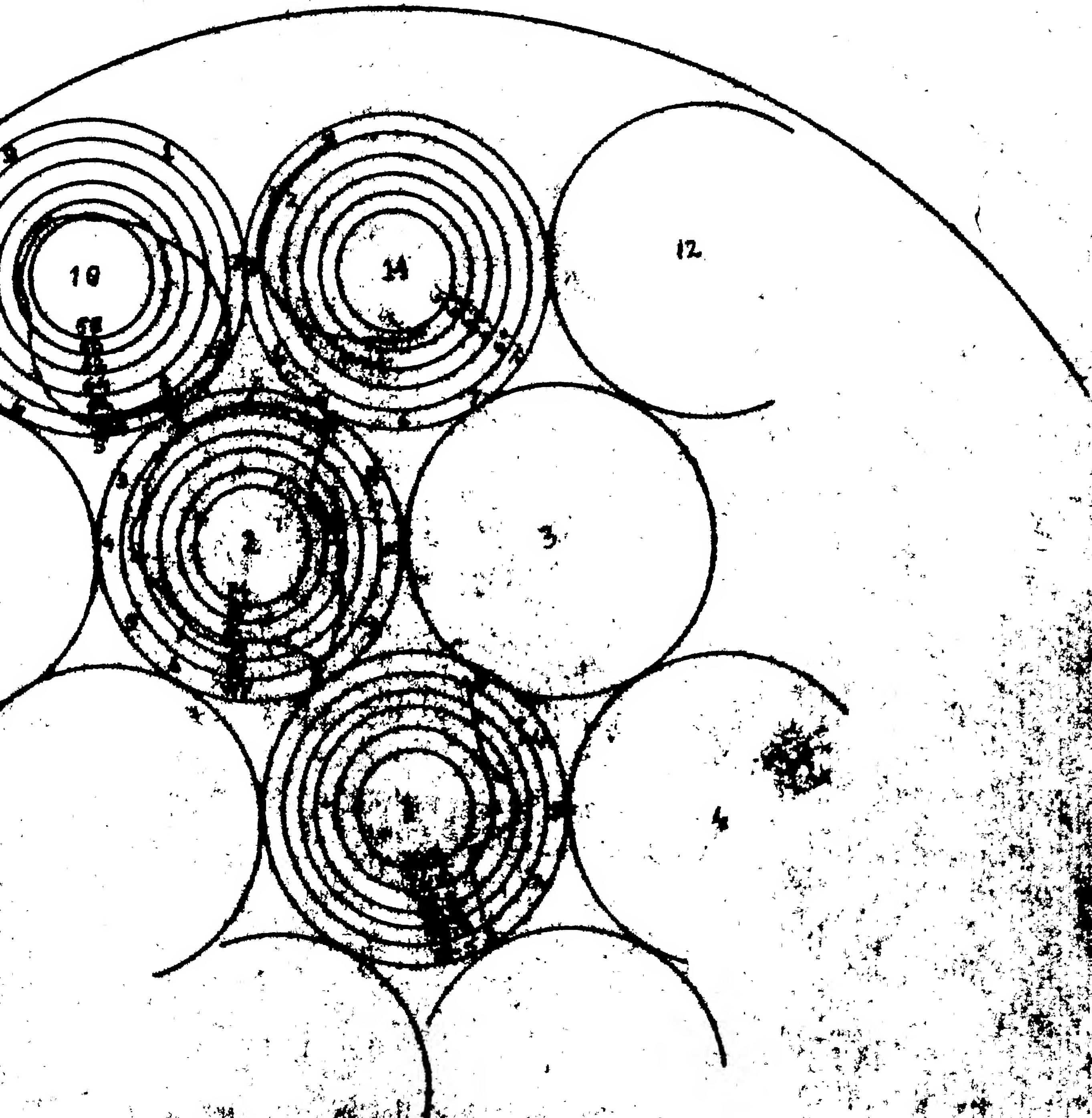
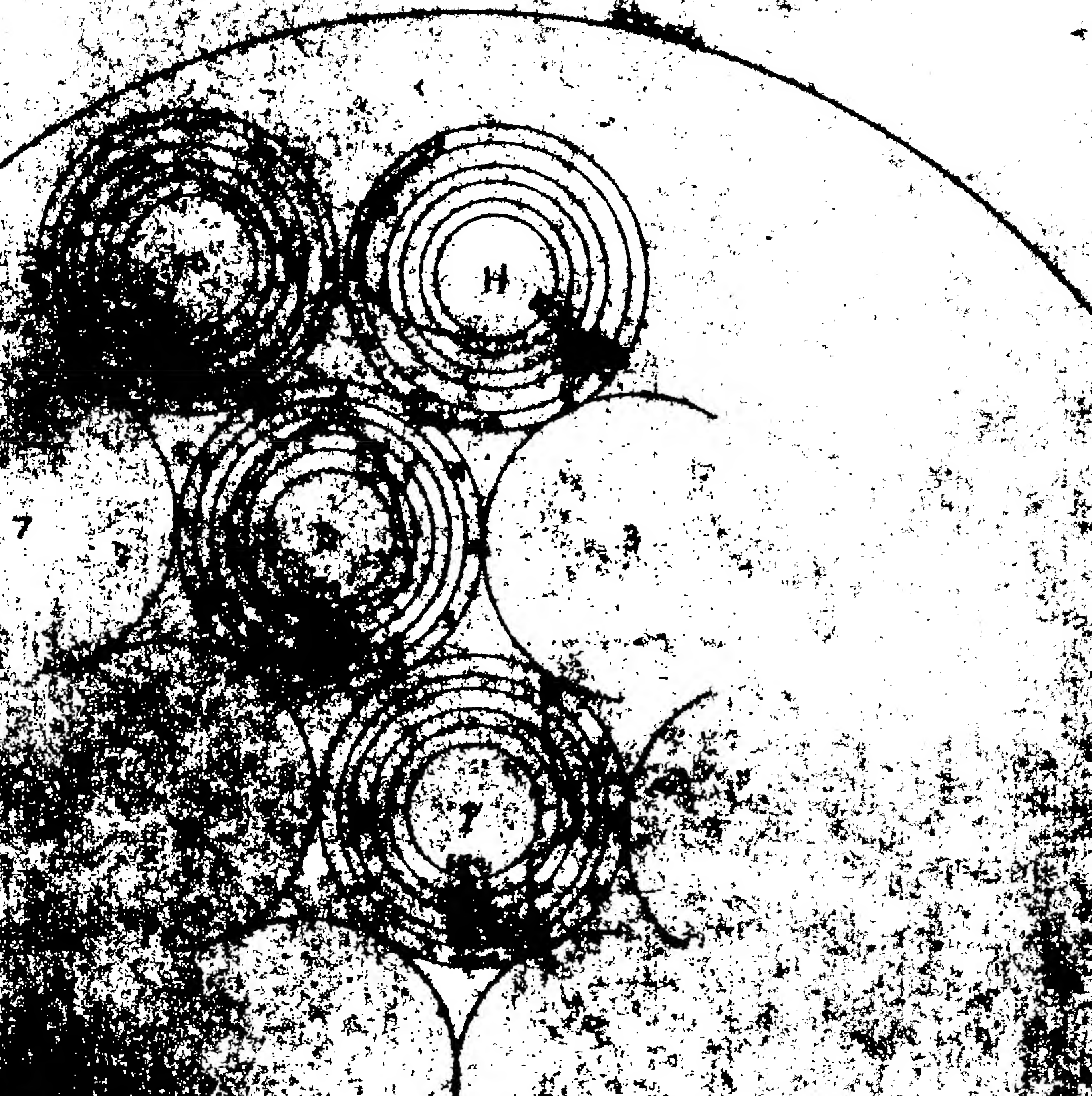


FIG. 16 TEMPERATURE DISTRIBUTION
[illegible] ([illegible])
[illegible] [illegible]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7
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